NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20034

WEAR STUDIES OF MIL-L-23699 AIRCRAFT TURBINE ENGINE SYNTHETIC BASE LUBRICATING OILS - 1 THE DEVELOPMENT OF A PROCEDURE AND INTIAL FINDINGS

By
J. C. Limpert

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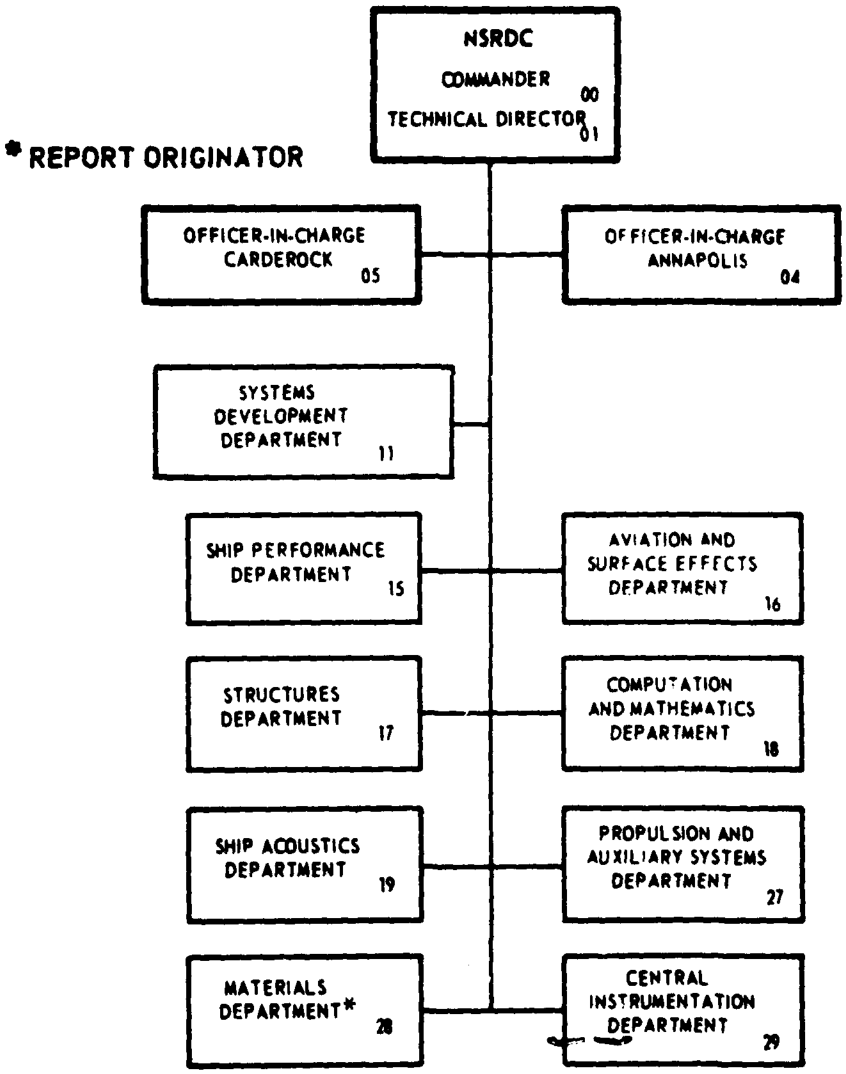

DEPARTMENT OF THE NAVY
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WEAR STUDIES OF MIL-L-23699 AIRCRAFT TURBINE ENGINE SYNTHETIC BASE LUBRICATING OILS - I - THE DEVELOPMENT OF A PROCEDURE AND INITIAL FINDINGS

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ABSTRACT

A new procedure has been developed to more easily distinguish differences in wear characteristics attributable to base-stock composition or additives and contaminants in MIL-L-23699 aircraft turbine engine synthetic base lubricating oils. The procedure employs a linear ball-on-flat principle and is capable of giving boundary lubrication wear information in terms of the work required to remove material from rubbing surfaces in the elastic range of bulk bearing material. The procedure is particularly useful in illuminating the initial stage of the wear process. The procedure is more sensitive in the low-wear region than a four-ball wear procedure used in steam-turbine oil specifications. It is capable of showing the detrimental effect on wear of water in gas-turbine lubricating oils.

The MIL-L-23699 oils exhibit a linear relationship between the removal of bearing surface material (wear) and the work required to effect this removal. This relationship is altered in the presence of at least one sea-water rust inhibitor. The MIL-L-23699 qualified oils vary widely in antiwear performance.

ADMINISTRATIVE INFORMATION

This report is for marinized gas turbine oils, milestone 2 for fiscal year 1973, of the Lubricating Fluids Program. This program, which is being conducted at the Annapolis Laboratory of the Naval Ship Research and Development Center, is described in the Program Summary of 1 May 1973. Task Area SF 54-543-501, Task 12410, and work unit 1-2831-126. Mr. E. A. Bukzin, NAVSHIPS (SHIPS 03421), is program manager, and Mr. R. P. Layne, NAVSEC (SEC 6101F), is the technical agent for the program.

At the Annapolis Laboratory, the work on the program falls under the cognizance of the Applied Chemistry Division (Dr. George Bosmajian) of the Materials Department (Mr. R. J. Wolfe, acting). The research is performed by the Fuels and Lubricants Branch (Mr. N. Glassman) with Mr. J. W. MacDonald as principal investigator.

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INTRODUCTION

The broadened application of the gas-turbine engine to ship propulsion poses potential problems in the development of suitable lubricants. While considerable progress has been made in "marinizing" the aircraft engines by such approaches as developing better salt-water-resistant turbine- and compressor-blade alloys, relatively little has been done with respect to marinizing the lubricants for these engines. The operational mode of the marinized gas turbine is considerably different from that of the aircraft gas turbine. The "start-stop" changing load-type of naval marinized gas-turbine operation would tend to maximize initial (break-in) wear of machinery components. In addition, this mode of operation also would produce the conditions most favorable for the condensation of water in the oil, although a prima facie case has not yet been established to support the likelihood of sea-water contamination entering the engine lubricating oil system. To date, contact with gas-turbine engine manufacturers and aircraft gas-turbine engine users has resulted in their pointing out that the possibility for such contamination exists.[1,2,3] Earlier NAVSHIPS,[4] under a contract, developed a sea-water rust inhibitor for use in MIL-L-23699 synthetic base aircraft gas-turbine oils. It was predicted by the contractor that the inhibitor system developed would not be suitable for use in all the various neopentyl polyol esters or pentaerythretol esters from which MIL-L-23699 oils (5.0 to 5.5 cSt at 210° F)* are formulated. This lack of general suitability of the sea-water rust inhibitor system in various qualified MIL-L-23699 oils was recently demonstrated.[5] The Naval Air Propulsion Test Center (NAPTC) recently conducted a full-scale, 300-hour, gas-turbine test of an oil containing the contract-developed rust inhibitor. The objective of the engine test was to determine whether the rust-inhibited oil would be detrimental to the engine. In that test, no detriment to the engine was detected, but the used oil lost its ability to inhibit rust.[6] This was confirmed at this laboratory.[7] The significance of these findings is two fold:

- Rust inhibitor systems suitable to a wider variety of ester base stocks used for MIL-L-23699 oils must be developed.

- Tests of fluids in gas-turbine engines to confirm the suitability of inhibited fluids will be required.

The present cost of full-scale tests in gas-turbine engines is approximately $75,000. This high cost, together with the expected number of rust-inhibited oil formulations to be evaluated, is sufficient incentive for research to find an alternate approach to engine testing, if indeed, improved marinized gas-turbine oils are to be obtained. One of the important things learned about lubricants in engine tests is their effectiveness in preventing engine wear. Thus, one of the alternate approaches to engine testing all rust-inhibited gas-turbine oil formulations is a bench test of sufficient significance to reduce the number of engine tests necessary. Candidate formulations would be required to perform satisfactorily in the relatively less costly bench test as a screening device. It has been shown[5] that six of the currently available MIL-L-23699 gas-turbine oils produce a wide range of wear rates (wear-scar diameters from 0.26 to 0.75 mm) when tested by a four-ball procedure that is used to evaluate steam-turbine lubricating oils.[8] Since all these oils were qualified by performing satisfactorily in a full-scale gas-turbine test, the value of that four-ball procedure can be challenged.

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[1] Superscripts refer to similarly numbered entries in the Technical References at the end of the text.

*Abbreviations, used in this text are from the GOP style manual, 1973, unless otherwise noted.

An additional factor that must be considered is how the oils would perform in an engine after sea-water contamination of the oil. This factor would require either both a dry and a wet engine test on every formulation made or engine tests on a reduced number of candidate formulations that had been selected on the basis of significant bench tests. Previous work at this laboratory,[9] using both time- and load-dependent four-ball tests, have failed to show a significant difference in results on a MIL-L-23699 gas-turbine oil when water was present. Conversely, other work in the presence of water and oxygen revealed a difference in performance in the four-ball sliding wear test between wet and dry low-viscosity (2.5 to 5.3 cSt) [illegible] at 210° F (99°C).[10] Thus, one aspect of solving the problem of obtaining suitable marine gas-turbine lubricating oils is the availability of a significant screening wear-test procedure. This is the subject of this report.

APPROACH

What is desired in the selection of a significant oil wear-test procedure is to relate the outcomes to service performance. Service tests are prolonged and difficult to control.[11]

The conditions to be found in the field are a combination of many variables operating simultaneously, making the simulation of these complex phenomena of friction and wear a difficult task. Friction and wear testing therefore is not regarded as an end in itself but rather as a means to an end i.e., reducing the number of candidate lubricants to be subjected to service evaluations.

It was decided after reviewing the American Society for Lubrication Engineers' summary of wear test[12] and on the basis of the author's experience,[13] to develop and use a wear test employing a ball-on-flat procedure and to conduct experiments at a load just below the plastic-deformation point of the bulk metal. Inferences about lubricant effectiveness in terms of volume of metal removed (wear) versus the work required to effect the removal could then be made.

It was further decided to conduct the initial study with balls of Rockwell hardness C63, 52100 bearing steel, ½ inch (12.7 mm) in diameter, sliding on a steel flat of similar properties.

METHOD

DETERMINATION OF LOAD

The objective is to establish a maximum load wherein mechanical wear is the result of displacement of surface asperities without bulk plastic deformation of metal contributing to the estimate of material removed.

For a ½-inch (12.7 mm) diameter (d) steel ball sliding on a steel flat, both of Rockwell hardness C63, the Poisson's ratio (v) is ≅ 1/3, the Young's modulus of elasticity (E) = 30 X 10^6 lb/in^2 (2.1 x 10^6 kg/cm^2), the proportional limit is ≅ 19.5 x 10^4 lb/in^2 (1.38 x 10^4 kg/cm^2), and the approximate shear yield stress is ≅ 150 x 10^3 lb/in^2 (10.5 x 10^3 kg/cm^2).

The procedure is based on the premise that the load should not exceed the elastic capabilities of the steel; therefore the maximum compressive stress in lb/in^2 (equation (1)),[14]

$$0.616 \sqrt[3]{\frac{WE^2}{[illegible]}} , \qquad (1)$$

where W is the load in pounds, may not exceed the proportional limit. In solving the equation

$$0.616 \sqrt[3]{\frac{WE^2}{d}} = 19.5 \times 10^4,$$

$$W = 8.82 \text{ lb} \cong 4 \text{ kg}$$

is obtained.

The procedure also is based on the premise that the load should not produce a maximum shear stress in excess of the shear yield stress of the steel; therefore, the maximum shear yield stress in lb/in^2 is obtained from the following:[15]

$$q_0 \sqrt{\frac{1}{16}(1 - 2v)^2 + \mu^2} \qquad (2)$$

where q_0 is in this case, the proportional limit and μ is the approximate coefficient of friction. ($\mu_{dry} \cong 2/3$, $\mu_{lubricated} \cong 1/10$). Solving this equation produces approximately 131,000 lb/in^2 (9150 kg/cm^2) for dry conditions and 25,000 lb/in^2 (1760 kg/cm^2) for lubricated conditions, both values being less than the approximate shear yield stress.

The approximate maximum subsurface shear stress is about one-third the proportional limit[15] and amounts to approximately 60,000 lb/in^2 (4210 kg/cm^2) which is less than the approximate shear yield stress.

It can be seen from the above calculations that the 4 kg load satisfies the desired conditions of conducting the wear experiments at a load just below the bulk plastic-deformation level of the wear specimens. This load, 4 kg, is responsible for the size of the initial wear scar on the ball (Hertzian diameter).

APPARATUS

For the initial wear studies, it was decided to apply the ball-on-flat approach in a linear concept by utilizing an available Instron Universal Testing Machine (IUTM) (figure 1) as a convenient way to obtain precisely controlled measureable motion capabilities together with existing load-measuring and recording capabilities. The IUTM was used to develop a unidirectional, cyclical motion of a ball moving on a stationary flat surface.

The ball is ½-inch (12.7mm) diameter 52100 steel. The flat is of similar composition. The ball is held at the lower extremity of a pivoted arm which is attached to the movable (10 cm/min maximum) crosshead of the IUTM (figure 2). At 10 cm/min lubrication is of the boundary type. The pivoted arm is deadweighted (4 kg) so that the ball is perpendicularly pressed against the vertical flat on the downstroke. On the upstroke, the ball is raised from the flat by a cam arrangement. The flat is held at the lower end of a tube (for low inertia), which is supported against the ball load but free floating otherwise. The upper end of the tube is attached to a load-measuring strain gage whose output is fed into a strip chart recorder. The recorder is active on the downstroke and inactive on the upstroke through a microswitch arrangement. The arm holding the ball and the tube holding the flat extend into a beaker containing the lubricating oil under study. The beaker is supported by a magnetic stirrer hot plate which maintains the lubricant at test temperature (80° ± 0.5°C). A permanent magnet is placed outside the beaker to attract free-floating wear particles.

PREPARATION OF SPECIMENS AND EXPOSURE TO PROCEDURE

The flat is polished to a finish comparable to that of the bearing ball. As illustrated in figure 3, this finish has a roughness variation of less the 0.1 μm. The bearing balls are already finished as received. The flat is polished by hand with 0.3 μm aluminum oxide powder wet with hexadecane (cetane). To minimize variation, the same ball and flat are used for a series of tests. The ball is positioned to expose a fresh surface each time, and the flat is moved laterally for each test. For "dry"* oil runs, the ball, the flat, and their holders are cleaned with naphtha and acetone and are air dried and installed in the apparatus. The test beaker and magnetic stirring bar are also cleaned with naphtha and acetone and air dried. The lubricating oil is added to the beaker and brought to temperature and cycling begun. A frictional trace is recorded at 1, 5, 10, 15, 20, 25, 30, 50, 75, 100, 125, 150, 200, and 300 cycles. At 1, 5, 15, 50, 100, 200, and 300 cycles the ball in its holder is removed and the wear scar optically measured, photographed, and surface roughness determined by Talysurf. At 300 cycles, the flat is also photographed and Talysurfed.

The wear track on the flat was usually visible as a slight discoloration but was never detectable in profile, figure 3. The wear track area on the flat is approximately 200 times the wear-scar area on the ball. Therefore, it was assumed that the consequences of the frictional load were applicable to the ball alone.

For runs with water in the lubricating oil, the procedure is identical to that for the dry oil runs except that ½% of distilled water is added to the oil in the beaker. The beaker is fitted with a conforming stainless steel closure (figure 4) containing a saturated argon bubbler in series with two water scrubbing towers (one internal in the oil and one external to the beaker) to humidify the argon. The steel lid is slotted to provide for the up and down movement of the ball-holding arm. The ball-holding arm is fitted with a sliding fitting which keeps moisture from escaping when the holding arm moves away to keep the ball from contact with the flat on the upstroke.

PROCESSING DATA

The method used to process the data is illustrated by applying the procedure to oil code E. The data in table 1 were obtained at a load of 4 kg, a speed of 10 cm/min, a stroke length of 5 cm, and a temperature of 80° C. This is boundary *lubrication*.

The raw data consist of frictional force traces and optically determined average ball scar diameters. A set of typical frictional force traces (for oil code E) is shown in figure 5. They show a typical stick/slip pattern. The area under each trace is mechanically integrated to obtain the average forces shown in the figure and in table 1. (To save time, if visual examination of frictional traces show them to be reasonably similar, these are not integrated but assigned values based on measured traces.)

Progressive wear-scar growth for oil E is shown in figure 6. The measurement of the wear-scar diameter is made with a calibrated eyepiece microscope. This can be done quite accurately by going very slightly off focus, whereby the perimeter of the scar shows up vividly as shown in figure 7 which was made with oil code A.

Figure 3, which was made with oil code D,** shows longitudinal and transverse profiles of a wear-scar after 50 cycles. The depth scale is 135 times that of the planar scale, hence the distortion

*Oil as received; water not deliveratety added.
**See note at bottom of table 4 on page 9.

TABLE 1
RESULTS OF PROCEDURE APPLIED TO OIL CODE E

Cycle	Average Scar Dia.(d) mm	Scar[1] Area mm^2	Scar Vol.[2] $mm^3 \times 10^4$	Average Force g	Average Surface Shear Stress g/mm^2	Cumulative Average Force g	Cumulative work g-cm x 10^5
1	0.236	0.0438	0.295	241	5500	241	0.012
5	0.237	0.0441	0.298	404	9160	342	0.085
10		–	–	417	–	–	–
15	0.238	0.0446	0.302	416	9330	387	0.290
20	–	–	–	406	–	–	–
25	–	–	–	391	–	–	–
30	–	–	–	379	–	–	
50	0.239	0.0449	0.309	352	7830	385	0.963
75	–		–	340	–	–	
100	0.242	0.0460	0.321	331	7190	359	1.795
125	–	–	–	349	–	–	–
150	–	–	–	331	–	–	–
200	0.246	0.0476	0.345	311	6530	343	3.43
300	0.250	0.0491	0.366	315	6420	332	4.98

(1) $\pi d^2/4$

(2) $\pi/3\ [2(6.35)^3 - \sqrt{4(6.35)^2 - d^2}\ (6.35^2 + d^2/8)]$

The longitudinal trace shows the buildup at the leading edge. The transverse trace slows the elastic recovery and its symmetrical shape. It was found that at 100 cycles, the elastic recovery was barely discernable and therefore was neglected in volume computations. This is in consonance with the work of Fein[16] who showed that elastic deformation of balls under load is important in estimating wear volumes from wear-scar diameters obtained under large loads and becomes negligible at loads below the plastic-deformation point of the material.

The frictional force traces are mechanically integrated to obtain the average forces. These values for oil code E are plotted in figure 8 and the curve thus formed is successively mechanically integrated to get the cumulative average forces up to each point where scar diameters are measured; namely, 1, 5, 15, 50, 100, 200, and 300 cycles. These cumulative average forces, times the number of cycles, times the length of stroke yield the work done in g-cm. The volume of material displaced from the ball is calculated from the optical measurements of the ball wear-scar diameter. The work done (g-cm) is plotted against the volume of material displaced (mm^3) in figure 9. This plot produces an essentially straight line. The cotangent of the angle between the plotted line and the hertz line is a number which is the amount of work required to displace a given amount of material (g-cm/mm^3). This value is a direct measure of the lubricants's ability to prevent wear. The steeper the slope, the lower the number and the less efficient the lubricant.

CHARACTERISTICS OF THE PROCEDURE

The characteristics of the procedure are illustrated in this section by again referring to the behavior of oil code F, table 1. Examination of the average force data, as plotted in figure 8, indicate that with the exception of the earliest time period, the shape of the curve is similar to the friction curve reported by Hall[17] using a four-ball wear machine on fluids such as those of interest here. The average surface shear stress is obtained by dividing the average force by the area. These data are plotted versus the number of test cycles in figure 10. It is to be noted that the average surface shear stress peaks almost immediately, rapidly decrease and become asymptotic. The curve in figure 10 gives the fine structure of the surface stresses acting on the metal asperities to produce the "running in" phase of wear referred to by Feng[18] and Schatzberg.[19]

When the volume of material removed in the wear process is plotted against the work required, a straight line results. This is in agreement with the work of Podlaseck and Shen[13] who used a hemispherical pin-on-disk machine to measure friction forces involved in sliding of unlubricated metal systems. They used as a basis the volume of metal removed versus work but with a more complex method of data reduction.

It is to be noted that the 4 kg load, which was calculated to permit conduct of the wear experiments in a region just below the bulk plastic-deformation level of the test ball, is responsible for the size of the initial wear scar on the ball. (The size of this scar also is known as the Hertzian diameter.)

The Hertzian diameter for this load is independent of the lubricant, and it is for this reason that the curve in *figure* 9 originates from a point that is greater than zero. The Hertzian diameter in inches is obtained from the formula:[14]

$$2 \times 0.881 \sqrt[3]{\frac{Wd}{E}}$$

converting to the metric system,

$$0.14859 \sqrt[3]{W_{kg.}} = 0.236 \text{ mm.}$$

This diameter was repeatedly verified experimentally.

Repeatability of the procedure is good, thus for example, oil B gave ratings of 24.1 and 24.7 g-cm x $10^9/mm^3$, and oil M gave ratings of 63 and 67 g-cm x $10^9/mm^3$.

DISCUSSION OF RESULTS

COMPARATIVE WEAR OF MIL-L-23699 OILS

The linear wear procedure was applied to six MIL-L-23699 synthetic gas-turbine lubricating oils representing the base-stock materials used to prepare all of the qualified oils. The test conditions were a 4 kg normal load, a sliding speed of 10 cm/min, a stroke of 5 cm length, and a temperature of 80° C. The results of these experiments are shown in table 2 and in figure 11.

TABLE 2
WEAR PERFORMANCE OF MIL-L-23699 QUALIFIED OILS

Work = g-cm x 10^5
Volume = mm^3 x 10^{-4}

Cumulative Cycles	OIL CODE A		B		E		I		J		M	
	Work	Volume	Work	Volume	Work	Volume	Work	Volume	Work	Volume	Work	Volume
1	0.017	0.295	0.018	0.295	0.012	0.295	0.018	0.295	0.016	0.295	0.013	0.295
5	0.104	0.46	0.112	0.30	0.085	0.298	–	Shortened			-	-
15	0.344	0.55	0.351	0.31	0.290	0.302	–	Procedure			–	–
50	1.42	1.08	1.23	0.35	0.963	0.309	–	–		–	–	–
100	2.98	1.9	2.34	0.40	1.79	0.321	1.78	0.365	1.81	0.353	1.70	0.320
200	5.72	3.3	4.51	0.49	3.43	0.345	–	–	–	–	–	–
300	8.55	4.9	6.64	0.57	4.98	0.366	–	–	–	–	–	–
Rating: g-cm x 10^9 per mm^3	1.85		24.1		70.2		25.2		31.2		67.6	

It is apparent that the six oils exhibit a range of effectiveness in preventing wear of the 52100 steel test ball under the test conditions. Since all the oils meet the same specification, it is apparent that the present method alone cannot be used to define a satisfactory engine lubricant. If no other information was available, it is reasonable to select the oil with the highest rating with respect to keeping wear at the lowest possible level.

It is of interest to see how this procedure rates oils when compared to the four-ball procedure[8] used to select steam-turbine oils. Table 3 gives the wear-scar diameters obtained on these oils after 2 hours at a 15 kg load, speed of 600 r/min, and 80° C. For convenience, the linear wear-test rating of the oils is repeated in table 3. A composite plot of the two procedures is given in figure 12.

It is apparent that the two procedures are ranking the oils in approximately the same order. It is of interest to note that where the four-ball wear-scar diameter is low, better resolution of relative oil effectiveness is possible with the linear wear procedure.

Effect of Water in the Oil on Wear

A preliminary examination was made to determine the suitability of the procedure for showing the effect on wear of the presence of water in oil. A ½% of distilled water was added to the beaker containing oil E. The closure (figure 4) was fixed to the beaker; the moist argon bubbling

at 1.0 l/min was begun; the magnetic stirrer was activated; and the oil-water mixture in the beaker was brought up to test temperature (80° C). At this point the procedure was conducted as for dry oils.

TABLE 3

FOUR-BALL WEAR TEST OF QUALIFIED MIL-L-23699 GAS TURBINE OILS

Oil Code	Average Wear Scar Dia - mm	Linear Test Rating g-cm x $10^9/mm^3$
A	0.75	1.85
B	0.44	24.4
E	0.27	70.2
I	0.34	25.2
J	0.40	31.0
M	0.29	65.0

It was found that the wear rating of oil E dropped from 70 g-cm x $10^9/mm^3$ (dry) to 3.65 g-cm x $10^9/mm^3$ (wet). It is planned to examine the other qualified oils similarly.

EFFECT OF SEA-WATER RUST INHIBITORS ON WEAR PROCESS

A preliminary examination was made in the linear wear procedure of a sea-water rust inhibitor, code X, in two of the qualified oils, codes A and B. Code A oil contained 1.27% of rust inhibitor X and Code B oil contained 1.90% of rust inhibitor X. Different concentrations of X were used since the inhibitor was effective in preventing rust at these concentrations in each oil.[5] The runs were conducted in the dry (no water) condition. The results are presented in table 4 and in figure 13.

It is immediately of interest to note that a departure from linearity occurs after approximately 100 cycles of operation. This suggests that the rust inhibitor is producing a separate effect. In one case, oil A, an improvement in wear performance was noted while in the other case, oil B, a deterioration in wear performance was noted. This behavior also was obtained in the four-ball wear test.[5] Since the same rust inhibitor was used one can speculate on two aspects: the amount that was used or the interaction with the base oil in which it was used.

TABLE 4

WEAR BEHAVIOR OF TWO RUST-INHIBITED GAS TURBINE OILS

Work = g-cm x 10^5 Volume = mm^3 x 10^{-4}				
Cumulative Cycles	Code C oil		Code D oil	
	Work	Volume	Work	Volume
1	0.016	0.295	0.016	0.295
5	0.112	0.366	0.112	0.367
15	0.35	0.44	0.449	0.457
50	1.18	0.55	1.48	0.86
100	2.45	0.707	3.12	1.34
200	4.82	0.76	6.38	2.02
300	7.12	0.80	9.41	2.18
300 cycle rating g-cm x 10^9 per mm^3	14.2		5.0	
Code C oil is code B oil plus rust inhibitor code X. Code D oil is code A oil plus rust inhibitor code X.				

ADDITIONAL VARIATIONS OF LINEAR WEAR PROCEDURE

An examination of figure 11 suggests that an early estimate of oil wear performance can be made when it is known that the oil will produce linear plots. Even when the oil wear departs from linearity, as in the case of the rust-inhibited oils, shown in figure 13, an early estimate of oil wear performance is possible, although with some risk of error.

Thus, an accelerated procedure would be as follows:

- Reduce the running time to 100 cycles (2 hours).
- Conduct all the calculations on the basis of 100 cycles.

To confirm this, an experiment using this shortened procedure was made with oil B. A rating of 24.1 g-cm x $10^9/mm^3$ was obtained in a 300-cycle test and a rating of 24.7 g-cm x $10^9/mm^3$ was obtained in a 100-cycle test.

CONCLUSIONS

- A linear ball-on-flat lubricating oil wear test has been developed which is capable of giving boundary lubrication wear information in terms of the work required to remove material from rubbing surfaces in the bulk bearing-material elastic range.

- The procedure is particularly useful in illuminating the initial stage of the wear process.

- The MIL-L-23699 oils exhibit linear relationships between wear and work required. This relationship is altered in the presence of at least one sea-water rust inhibitor.

- The MIL-L-23699 qualified oils vary widely in antiwear performance.

- The procedure is more sensitive in the low-wear region than a four-ball wear procedure used in steam-turbine oil specification.

- The procedure is capable of showing the determintal effect on wear of water in gas-turbine lubricating oils.

FUTURE WORK

It is planned to extend the use experience with the linear wear procedure as follows:

- Explore the variables of speed, temperature, and other loads.
- The effect of water in all of the qualified oils.
- The effect of other proposed sea-water rust inhibitors.
- The wear rates of other bearing materials.

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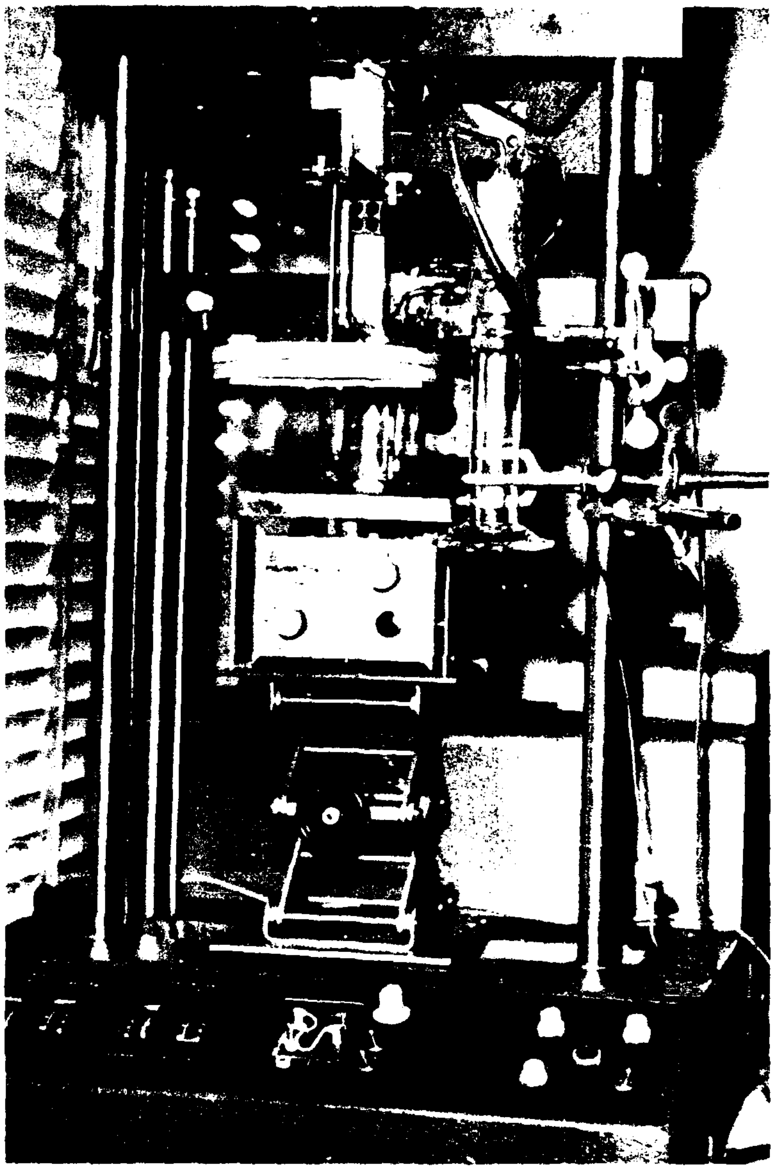

Figure 1
General View of Wear Test Apparatus

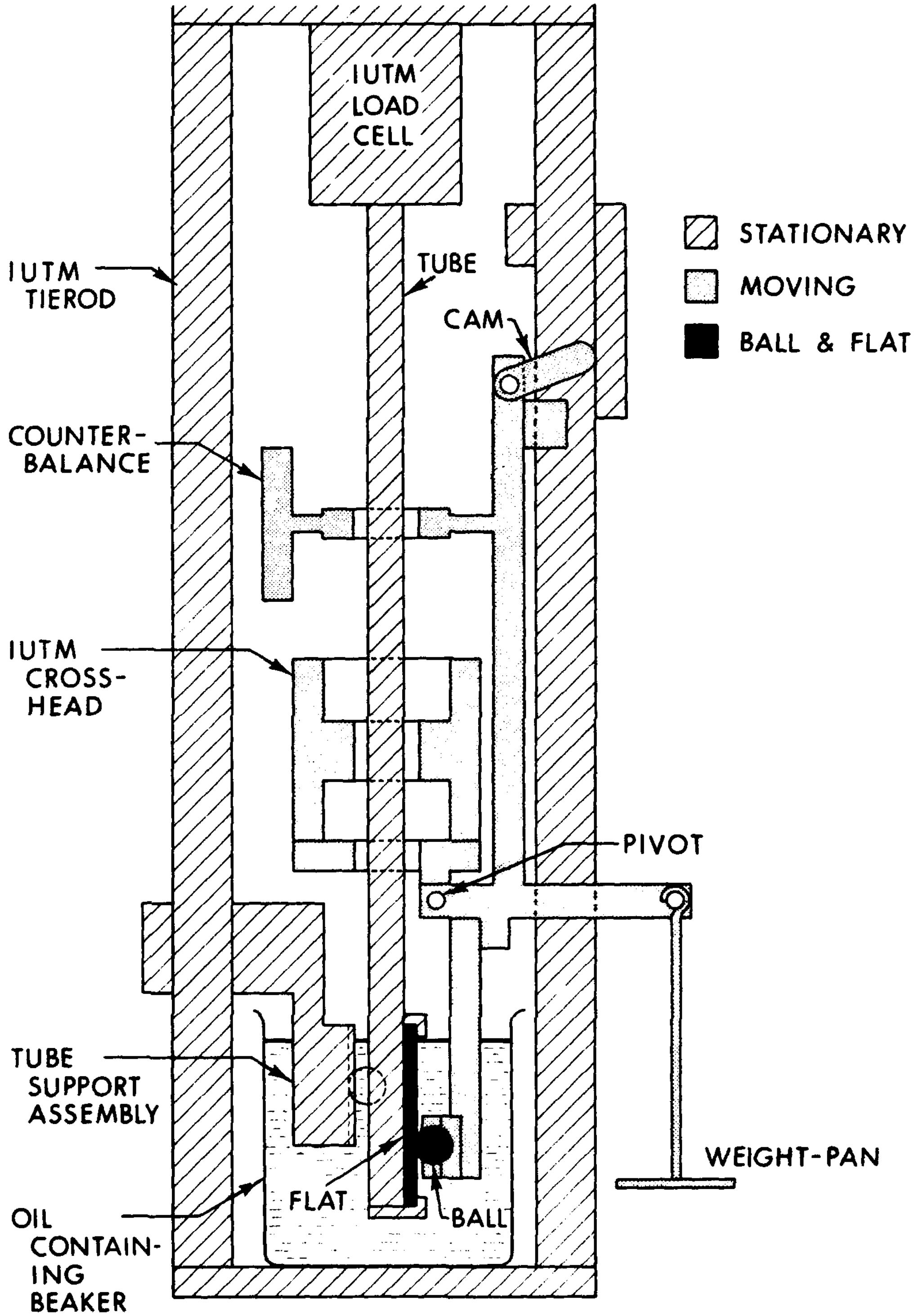

Figure 2
Linear Wear Test Apparatus

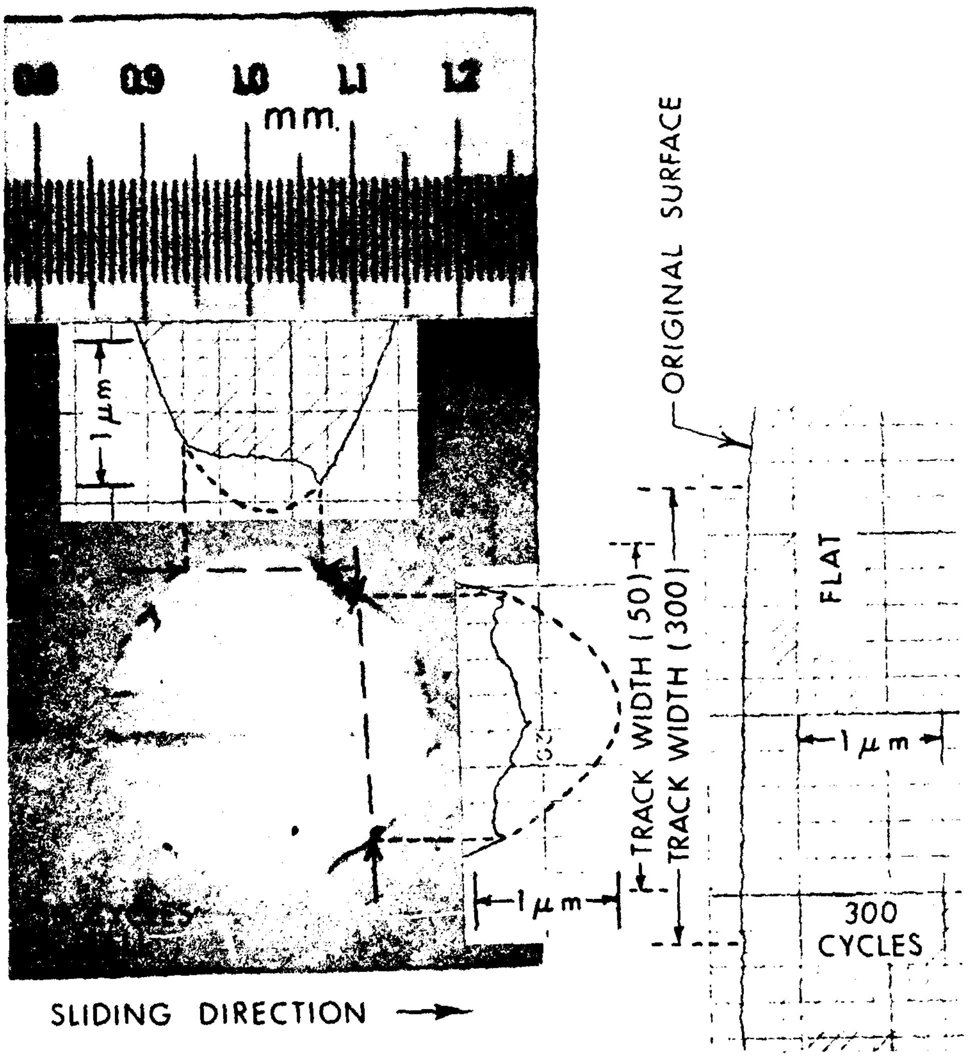

Figure 3
Surface Finish and Profiles of Ball
and Flat (by Talysurf)

Figure 4
Exploded View Showing Details
of Moisture Retaining Closure

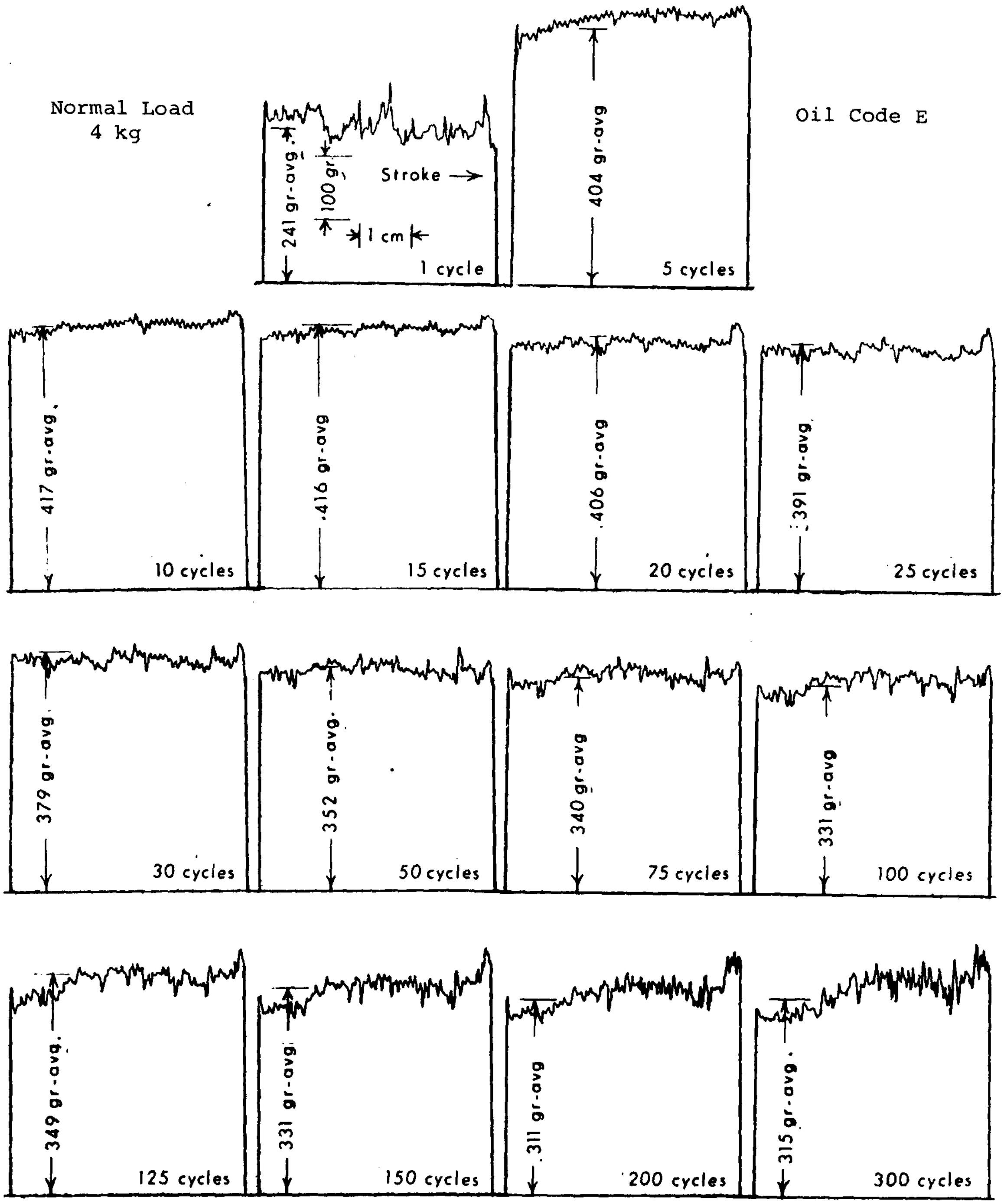

Figure 5
Frictional Force Traces
Oil Code E

OIL CODE E
(Arrows Denote Sliding Direction)

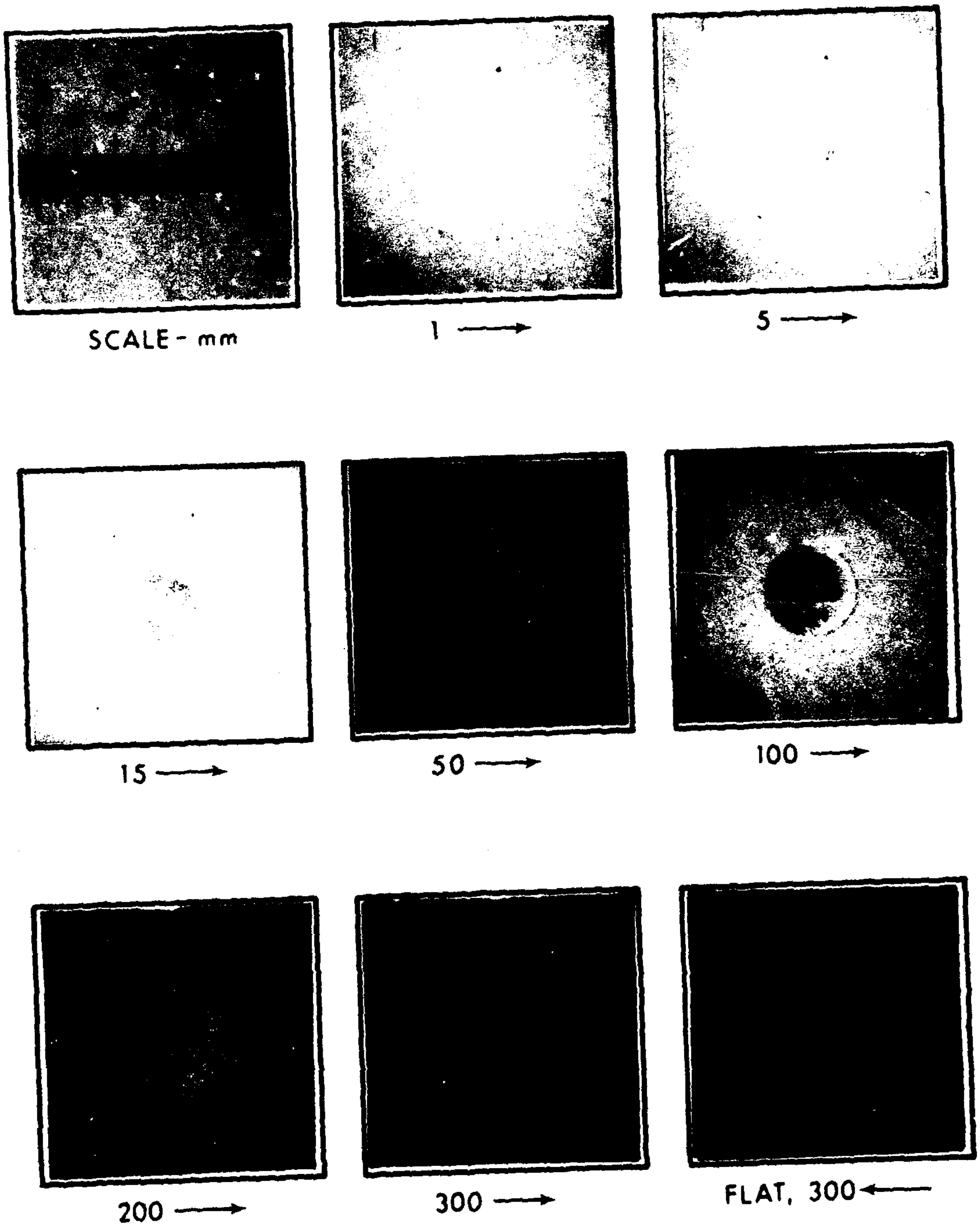

Figure 6
Wear Scar Growth Cycles

OIL CODE A
(Arrows Denote Sliding Direction)

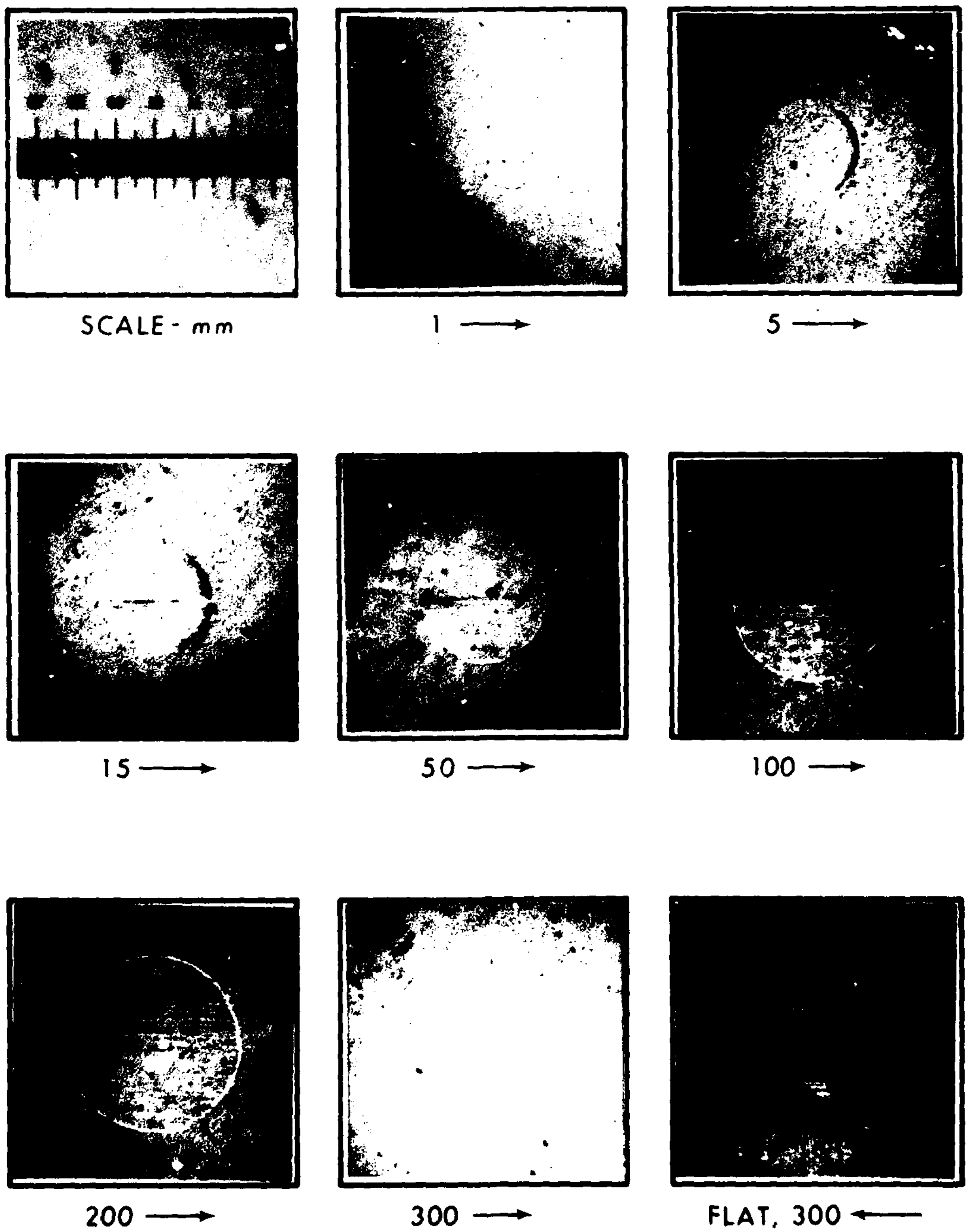

Figure 7
Wear Scar Growth Cycles

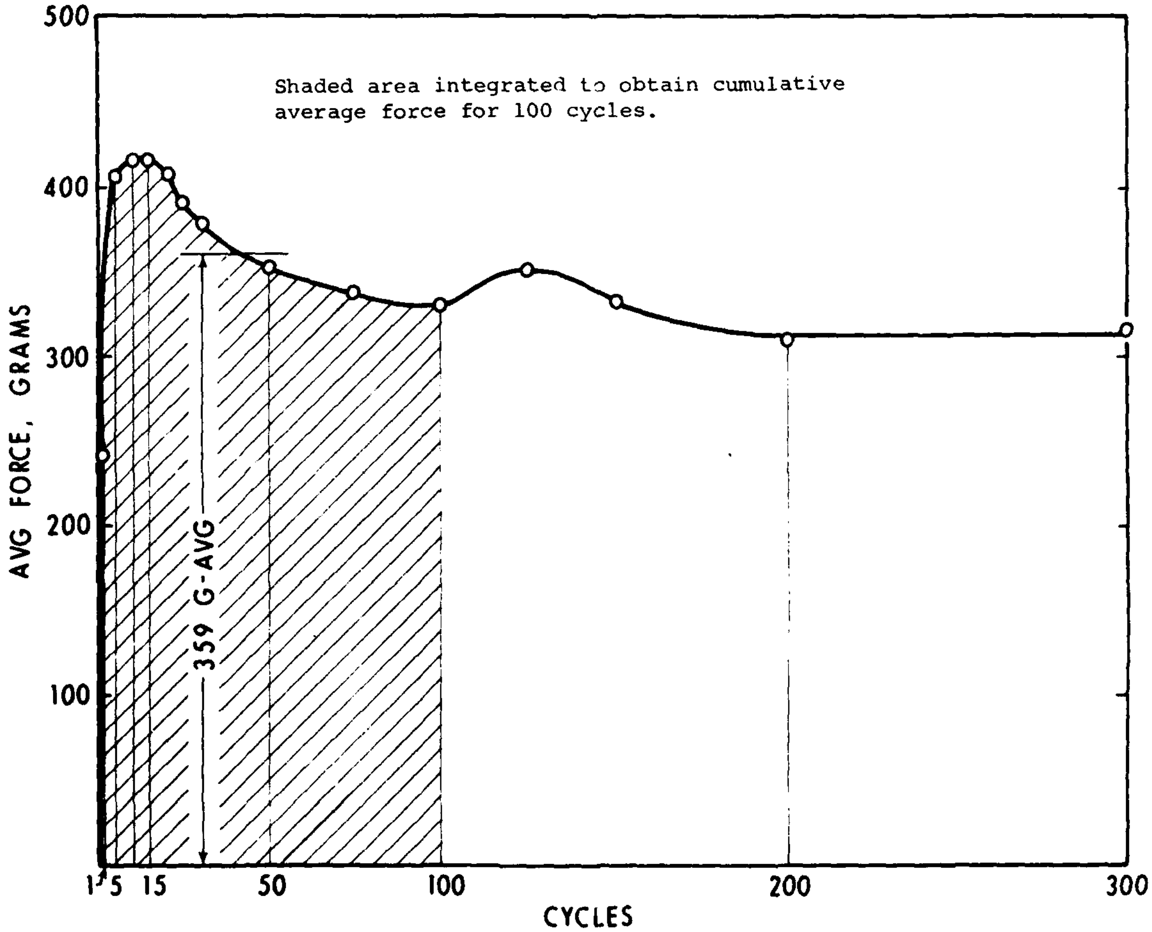

Figure 8
Average Force Curve, Oil Code E

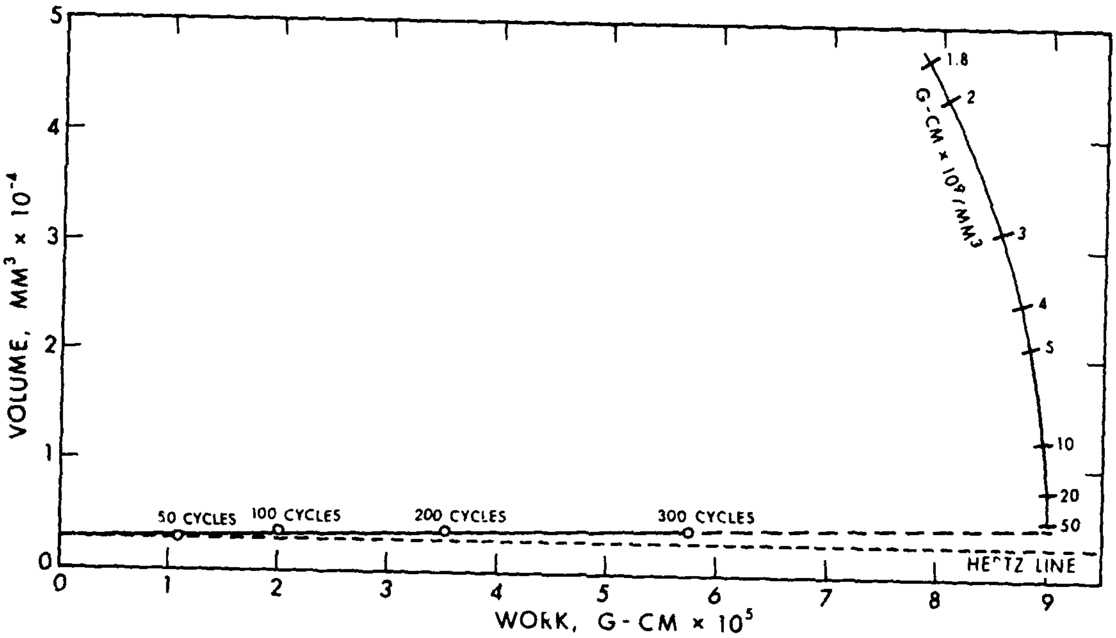

Figure 9
Work Versus Volume of Material
Displaced, Oil Code E

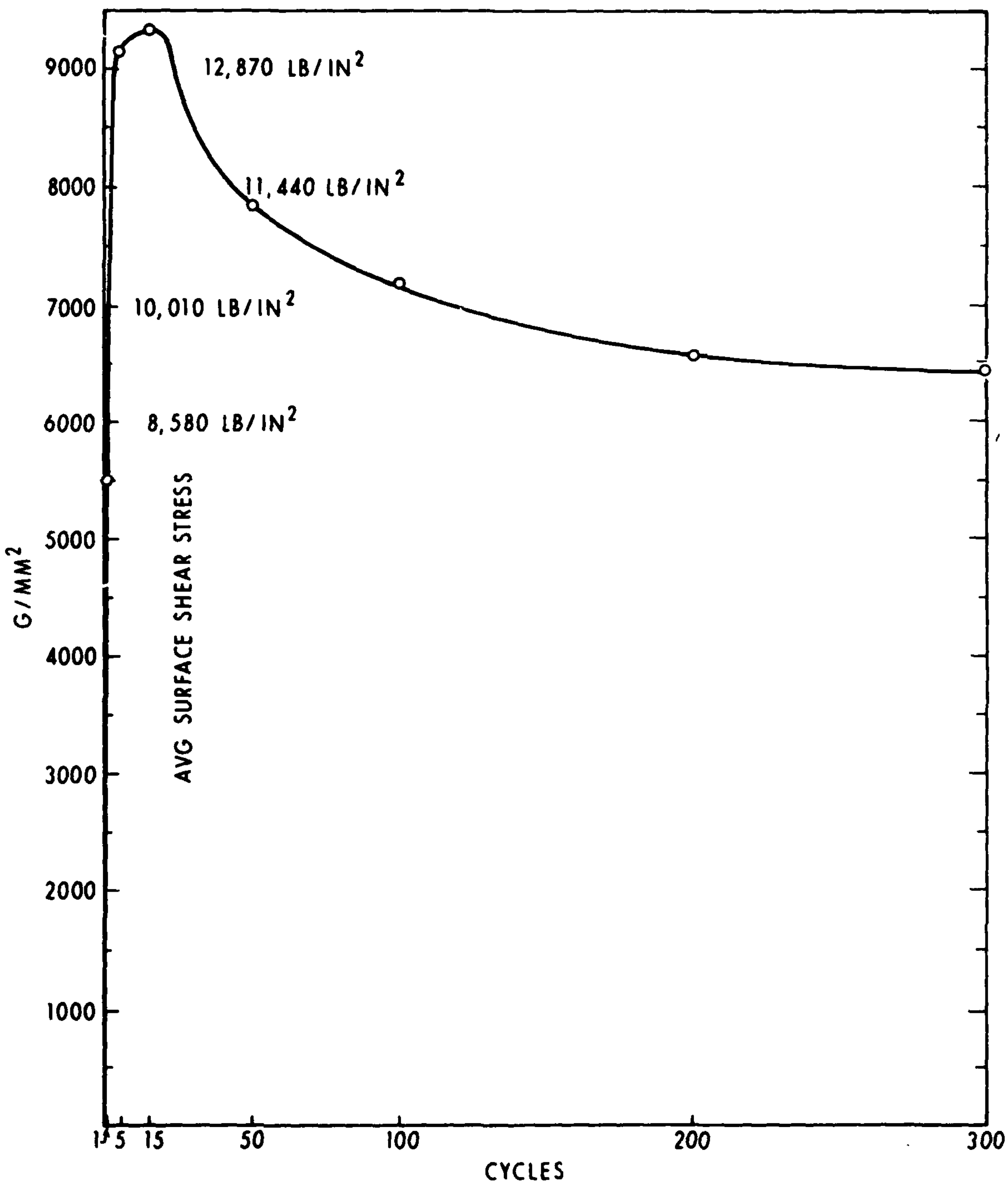

Figure 10
Surface Shear Stress, Oil Code E

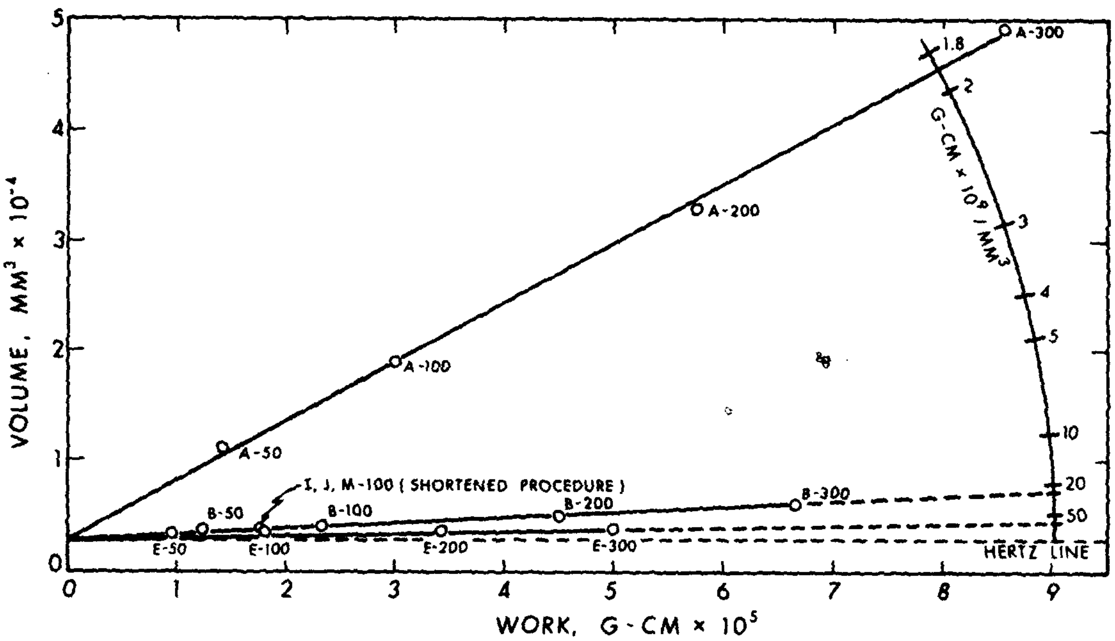

Figure 11
Comparative Wear of Qualified
MIL-L-23699 Oils

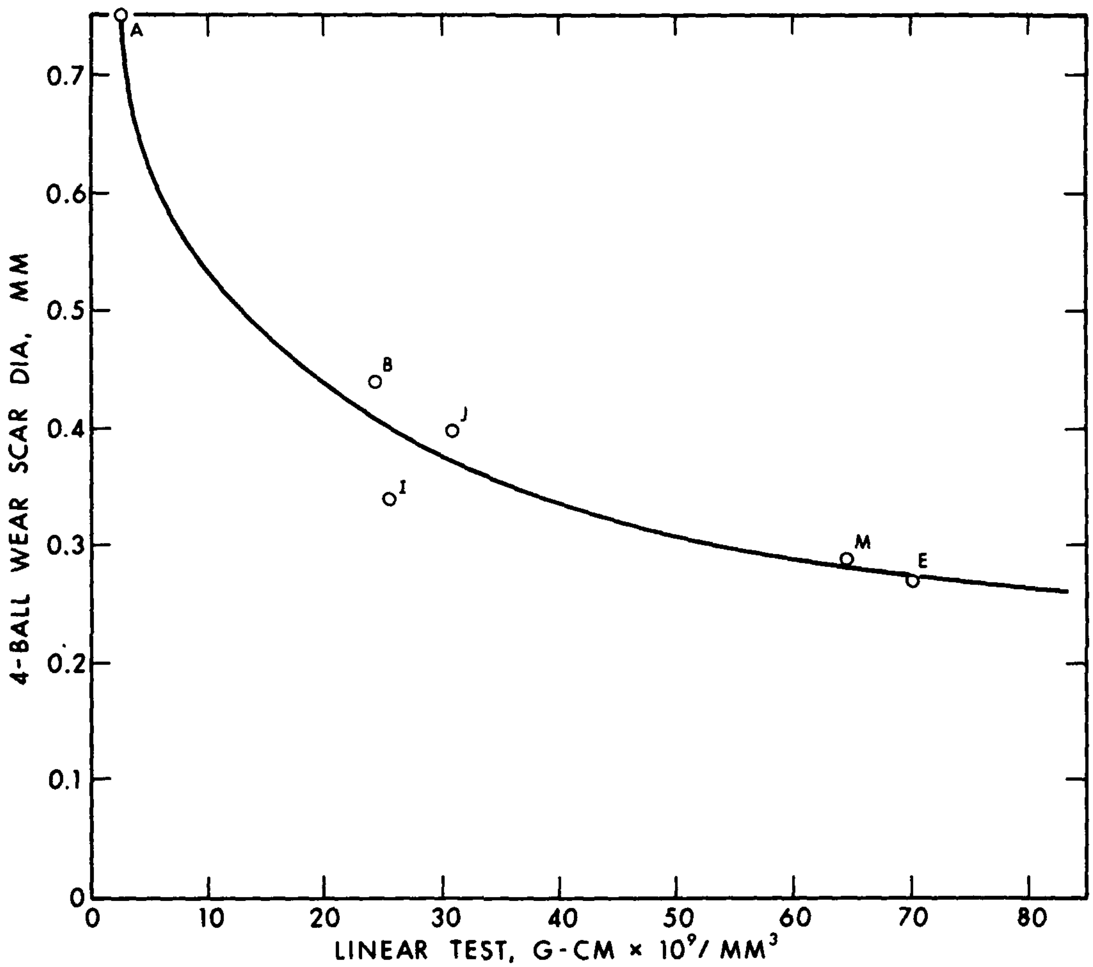

Figure 12
Comparison of Two Wear Procedures

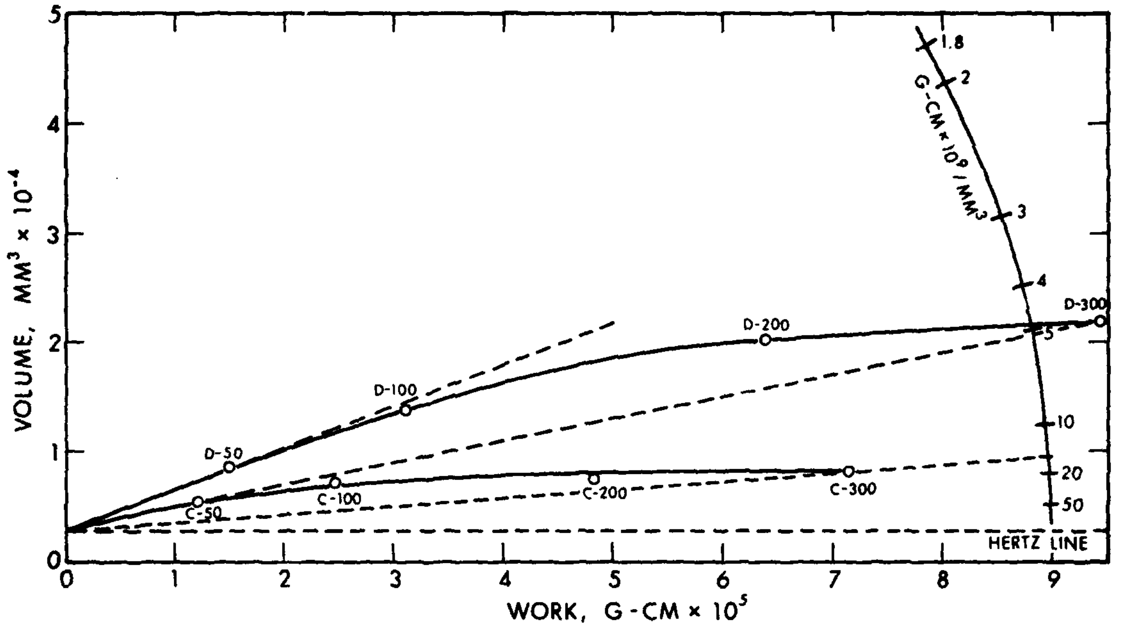

Figure 13
Effect on Wear of Rust Inhibiting
Two Gas Turbine Oils

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13. ABSTRACT

A new procedure has been developed to more easily distinguish differences in wear characteristics attributable to base-stock composition or additives and contaminants in MIL-L-23699 aircraft turbine engine synthetic base lubricating oils. The procedure employs a linear ball-on-flat principle and is capable of giving boundary lubrication wear information in terms of the work required to remove material from rubbing surfaces in the elastic range of bulk bearing material. The procedure is particularly useful in illuminating the initial stage of the wear process. The procedure is more sensitive in the low-wear region than a four-ball wear procedure used in steam-turbine oil specifications. It is capable of showing the detrimental effect on wear of water in gas-turbine lubricating oils.

The MIL-L-23699 oils exhibit a linear relationship between the removal of bearing surface material (wear) and the work required to effect this removal. This relationship is altered in the presence of at least one sea-water rust inhibitor. The MIL-L-23699 qualified oils vary widely in antiwear performance.

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14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
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Synthetic oils						
Gas Turbines						
Marine						
Inhibitors						
Water contamination						

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